

Torque and power characteristics of a helium charged Stirling engine with a lever controlled displacer driving mechanism

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ABSTRACT

This study presents test results of a Stirling engine with a lever controlled displacer driving mechanism. Tests were conducted with helium and the working fluid was charged into the engine block. The engine was loaded by means of a prony type micro dynamometer. The heat was supplied by a liquefied petroleum gas (LPG) burner. The engine started to run at 118 °C hot end temperature and the systematic tests of the engine were conducted at 180 °C, 220 °C and 260 °C hot end external surface temperatures. During the test, cold end temperature was kept at 27 °C by means of water circulation. Variation of the shaft torque and power with respect to the charge pressure and hot end temperature were examined. The maximum torque and power were measured as 3.99 Nm and 183 W at 4 bars charge pressure and 260 °C hot end temperature. Maximum power corresponded to 600 rpm speed.

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1. Introduction

Stirling engine is a regenerative externally heated engine operating with a cycle that has the same thermal efficiency with Carnot cycle if it is ideal and lossless. Since it can be powered by various heat sources and is also able to use solar radiation and waste heat as energy source, it attracts attention of engineers working on renewable energy. Theoretical and experimental numerous studies have been conducted on Stirling engines. Many types have been designed and built [1,2] however; none of them became competitive with the internal combustion engines and its commercial usage is not at desired level. Leakage of working fluid out of the engine, larger volume and mass compared to internal combustion engines, longer response period to speed and power changing requirement, higher quality temperature resisting material requirements are some of the disadvantages of Stirling engines. Releasing low pollutant emissions, working silent, requiring less maintenance and ability of being powered by variety of energy sources are some of the useful features of Stirling engines [1–3].

Performance of Stirling engines depends on; the thermal and physical properties of working fluid, temperature difference between hot and cold ends, charge pressure of working fluid, effectiveness of heat exchangers used as regenerator, heater and cooler, approximation degree of practical cycle to theoretical cycle, minimization degree of flow and mechanical losses and so on [3].

As working fluid, Stirling engines use a compressible fluid such as; air, hydrogen, helium, nitrogen or even vapors. In general, hydrogen and helium are used because of their higher heat-transfer capabilities than other fluids. Hydrogen is thermodynamically a better choice, however not safe to work with. Helium is a noble gas and safer to use [1,4,5].

Despite of that the thermal efficiency of theoretical Stirling cycle is identical to Carnot, in practice due to irreversibilities caused by external heating and imperfect regeneration, thermal efficiency is lower than Carnot [6,7]. Principal sources of irreversibilities are temperature differences between hot source and fluid in the hot volume, cold source and fluid in the cold volume, finite capacity of hot and cold sources, flow frictions, leakage of working fluid during compression and expansion processes, etc.

To improve the specific power of Stirling engines, the mass of working fluid in the working space is increased. Increase of working fluid mass requires increase of the inner heat-transfer area or heat-transfer coefficient of the engine as well. Increase of the heat-transfer coefficient may be achieved by means of narrowing the flow channels of heat exchangers, however; this causes the flow losses to increase. Therefore increase of heat-transfer coefficient provides a partial resolution to the problem and requires an optimization. Increase of inner heat-transfer surface area makes the dead volume larger and causes the compression ratio to decrease and reduces the work generation. As a result, the specific power of Stirling engines is not competitive with internal combustion engines and its application scope is limited to stationary power generation plants. For the current situation the technology level of Stirling engines is eligible only for solar energy investigations and domestic cogeneration

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systems powered with natural gas. Within the last three decades, several Stirling-dish solar energy conversion systems were built and tested [8,9]. Most of them used free piston Stirling engines. Practical efficiencies, from sun radiation to electricity, exceeding 25%, and output powers rising to 50 kW were reported [10]. Domestic scale heat and electricity cogeneration by means of Stirling engine was a concept considered about two decades ago [11,12] and it remains still as a concept [13].

In Stirling engine investigations one of the recent developments is the concept of low temperature differential (LTD) Stirling engine. The LTD Stirling engine can be run with very small differences in temperature between the hot and cold ends of the displacer cylinder. Therefore, these engines are able to use low grade, cheap and waste heat sources including solar energy, flue gas, geothermal hot water, etc. Two configurations of LTD engines are the Ringbom and kinematic engines. In Ringbom engine, the power piston is connected to the flywheel and there is no kinematic connection to the displacer. In the kinematic engine, the displacer and the power piston are kinematically connected to the flywheel [2,14].

Abdullah et al. [15] described the critical parameters of the LTD Stirling engine design according to the Schmidt analysis and the third order analysis. Kongtragool and Wongwises [16] manufactured and tested gamma-configuration LTD Stirling engines with twin and four power pistons. Engines were tested with various heat inputs using an LPG fuelled gas burner. The engine produced 32.7 W output power at 771 K heater temperature and atmospheric pressure.

This research aims to develop a Stirling engine for solar energy and the other low and moderate temperature energy sources having 200–500 °C heating range. The research is being sponsored by The Scientific and Technological Research Council of Turkey (TUBITAK). The mechanical arrangements, kinematic and thermodynamic analysis of the engine were recently presented in Ref. [17]. The performance characteristics obtained with air were also presented in Ref. [18]. This paper presents performance characteristics of the engine obtained with helium. The systematic tests of the engine were conducted at 180 °C, 220 °C and 260 °C hot end external surface temperatures and the charge pressure was varied in the range of 1–4.5 bars.

2. Experimental setup

Construction details of the engine are given in Table 1 and illustrated in Fig. 1. Front and side photographs of the engine are shown in Fig. 2. The cylinder of the engine consists of two sections connected to each other end-to-end. The first part functions as piston liner and made of oil hardening steel. The second part functions as displacer cylinder and made of ASTM steel. The displacer cylinder inner heat-transfer surface was augmented by means of growing rectangular spanwise slots with 3 mm depth and

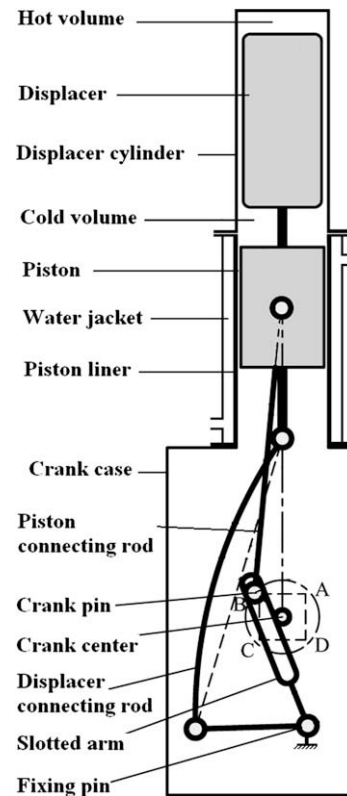


Fig. 1. Schematic view of the test engine.

2 mm width. The top surface of the displacer cylinder was also augmented by growing annular slots. The total heat-transfer surface of the engine is 0.1705 m². In the case of no slot, the inner surface would be 0.0796 m². These numbers does not include the surface of displacer which is 0.0527 m². In the engine no separate regenerator was used. Duty of regenerator is performed by the surface of displacer and displacer cylinder.

The inner surface of the piston liner was finished by honing. The external surface of the piston liner is cooled by means of a water circulation directly contacting to the external surface of the liner. The pistons were made of aluminum alloy because of its light weight and machining simplicity.

As shown in Fig. 1, the crankshaft has only one pin for the connection of piston rod and displacer driving mechanism. The motion of the displacer is governed by a lever. Karabulut et al. [17] compared the theoretical work generation of this engine with rhombic-drive and crank-driven β type engines and indicated its advantages. The lever consists of two arms with 70 degree conjunction angle. One of them holds a slot bearing as the other holds a circular bearing. At the corner of lever, a third bearing exists. The lever was mounted to the body of engine through the bearing at the corner by means of a pin. The slot bearing at one arm of the lever was connected to the crank pin. The circular bearing at the other arm of the lever was connected to the displacer rod. While the crank pin turns around the crank center, it drives the lever fort and back around the pin connecting the lever to the body. The other arm connected to the displacer rod moves the displacer up and down. Both ends of crankshaft were bedded with ball bearings. Escape of working fluid through the crankshaft beds was prevented by means of adapting an oil pool around the end pins of crankshaft. As long as the charge pressure remains below 5 bars, no working fluid leakage occurs. To lubricate components of the driving mechanism, the crankcase of the engine was filled with some oil.

Table 1
Technical specifications of the test engine.

Parameters	Specification
Engine type	β
Power piston	Diameter $\times$ stroke (mm) 70 $\times$ 60 Swept volume (cc) 230
Displacer	Diameter $\times$ stroke (mm) 69 $\times$ 79 Swept volume (cc) 295
Working fluid	Helium
Cooling system	Water cooled
Compression ratio	1.65
Total heat-transfer area of the displacer cylinder (cm ²)	1705
Maximum engine power	183 W (at 590 rpm)

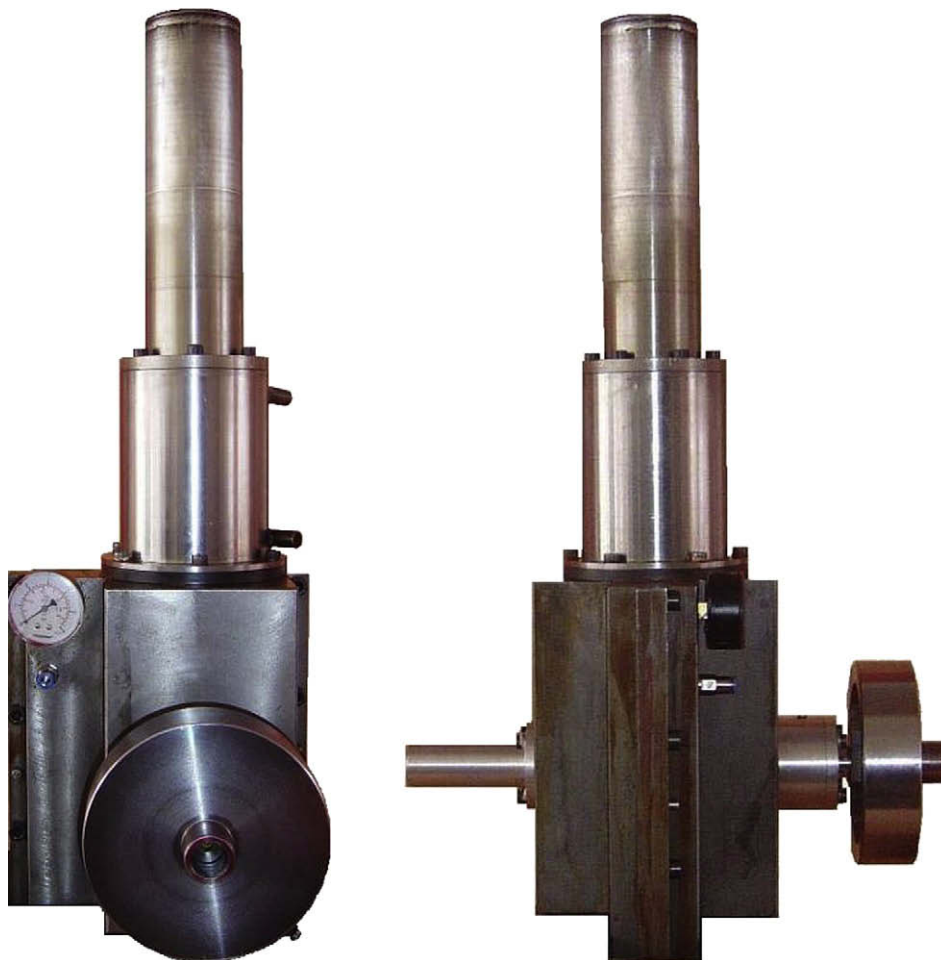


Fig. 2. Photographs of β -type Stirling engine (front and side views).

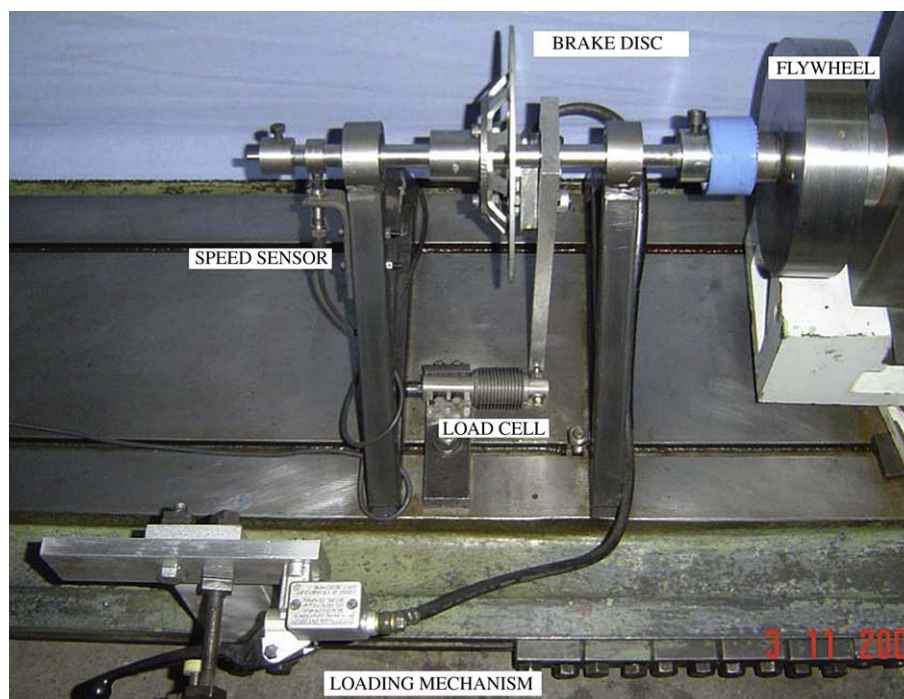


Fig. 3. Photograph of the experimental setup.

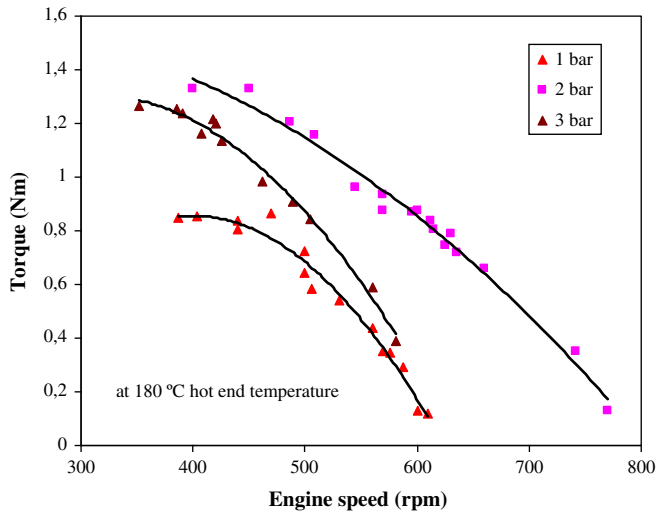


Fig. 4. Variation of engine torque versus the speed at 180 °C.

The engine was tested by means of a prony type micro dynamometer equipped with a load cell and a tachometer with ESIT brand names. The photograph of dynamometer is shown in Fig 3. The loading accuracy of dynamometer is 0.003 Nm. Temperature was measured with a non-contact infrared thermometer, DT-8859. The charge pressure was measured with a bourdon tube pressure gauge with 0.1 bar accuracy and 0–10 bars measurement range. A pressure regulation valve was used to adjust the charge pressure. The working fluid was charged into the block of the engine. The heat was supplied by an LPG burner.

3. Test procedure

Before conducting systematic tests, the engine was run under several working conditions and mechanical and thermal problems were eliminated. The working clearance of piston was experimentally optimized by running the engine under different hot end temperatures and charge pressures. The oil pool placed around the end pin of the crankshaft was checked up to 5 bars charge pressure and no leak was observed. Clearance between displacer and displacer cylinder was optimized, according to variation of the power, by reducing the diameter of the displacer with 0.1 mm steps.

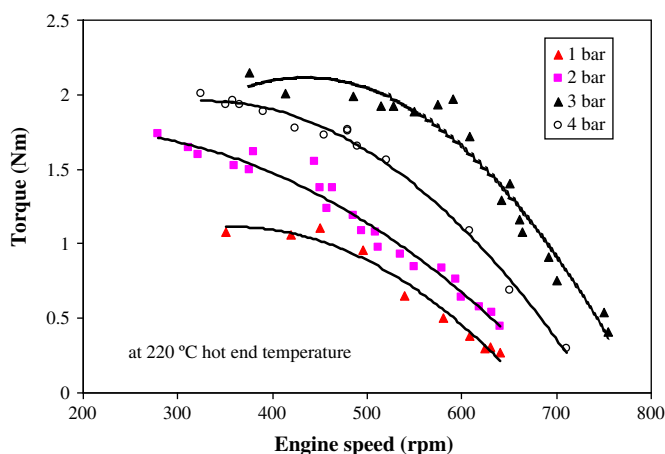


Fig. 5. Variation of engine torque versus the speed at 220 °C.

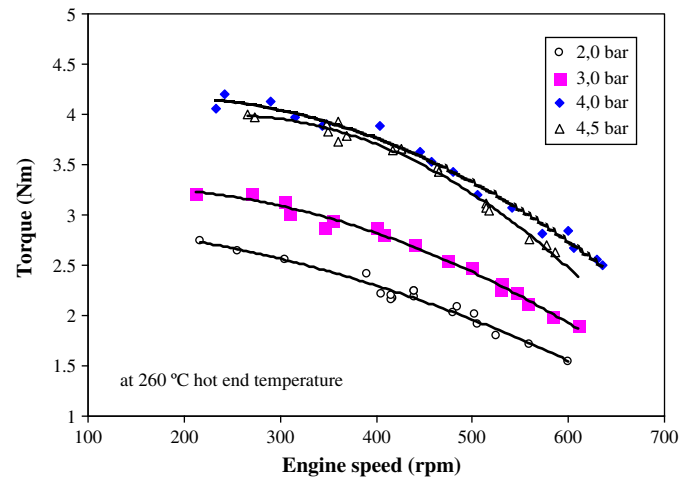


Fig. 6. Variation of engine torque versus the speed at 260 °C.

Test operation includes measurement of hot end temperature, cold end temperature, block pressure, engine speed and torque. The block pressure and cold end temperature do not require a continuous measurement. As testing operation progresses, the hot end temperature was steadily measured and its variation was limited by means of adjusting the distance between flame and hot end of the engine. As is known the torque produced by the engine is equal to the torque simultaneously generated by the external load. Since the speed and torque of the engine are related to each other, the torque was set to any desired value by adjusting the external load and the speed was measured. To ensure correctness of measurement, reading of speed was made after a couple of seconds than changing the external load.

The minimum hot end temperature for the engine to run with helium was measured as 118 °C. Experiments were conducted at 180 °C, 220 °C and 260 °C hot end temperatures. Since the external surface of the displacer cylinder was finless, hot end temperatures could not exceed 260 °C. At each value of the hot end temperature, the charge pressure was varied from 1 bar to a state that the power declines.

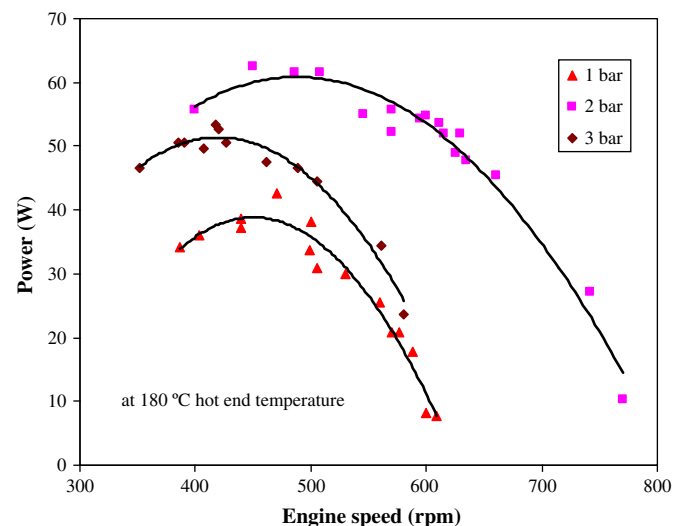


Fig. 7. Variation of engine power versus the speed at 180 °C.

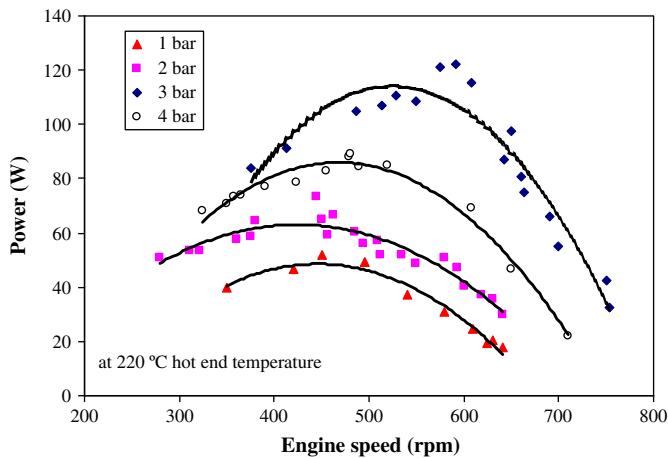


Fig. 8. Variation of engine power versus the speed at 220 °C.

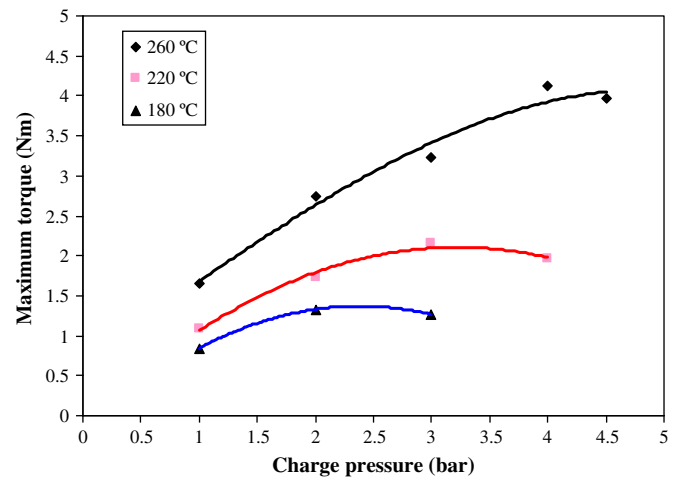


Fig. 10. Variation of maximum torque versus the charge pressure.

4. Results and discussions

Relation of the speed and torque was illustrated in Figs. 4–6 for 180 °C, 220 °C and 260 °C hot end temperatures. If the external load is gradually increased, as the torque of the engine increases the speed decreases up to a certain value and then the engine stops. Before the engine stops, the crankshaft speed exhibits larger oscillations. Therefore, data obtained for speeds near enough to stopping point are not reliable and left out of the evaluation. As the speed of the engine decreases, the torque increases but the slope of the speed–torque curves decrease. Therefore, increase of the torque ends at a certain value of speed and the engine stops.

Increasing heat exchange time and decreasing flow losses, which are results of decreasing speed, have positive effects on the torque. However, increase of the escaping working fluid, which is also a result of decreasing speed, has a negative effect [18]. At lower speeds, the negative effect of escaping fluid prevails the positive effects and the increase of torque terminates. The speed–torque curves obtained by Kongtragool et al. [16] and Tavakolpour et al. [19] have slopes always increasing with decreasing speeds. Running speeds of their engines were lower than the engine presented here more than 20 times. That may cause the difference of slope.

For each value of the hot end temperature applied to the engine, there appears an optimum charge pressure at which a higher speed–torque curve is obtained. For 180 °C, 220 °C and 260 °C hot

end temperatures optimum charge pressures are 2, 3 and 4 bars as seen in Figs. 4–6 respectively. Maximum torques corresponding to 180 °C, 220 °C and 260 °C hot end temperatures are 1.33 Nm, 2.15 Nm and 3.99 Nm. Corresponding speeds to these values of torque are 400 rpm, 376 rpm and 266 rpm. From Figs. 4 to 6, it is noted that as the hot end temperature increases, the resistibility of engine against the external load becomes better and can work at rather low speeds.

Relation of the speed and power was illustrated in Figs. 7–9. All of the speed–power curves have a maximum at a certain value of speed. The speed providing the maximum power varies with respect to the charge pressures and hot end temperatures, however, it appears in the range of speed; 500–600 rpm. When the engine was tested with air as working fluid, maximum shaft torque and power were determined as 1.17 Nm and 52 W respectively [18].

At constant hot end temperatures, variation of torque and power with respect to charge pressure are illustrated in Figs. 10 and 11. For each value of the hot end temperature we observe that the charge pressure has an optimum value. Fig. 12 illustrates variation of the power with respect to hot end temperature. All of the data used in Fig. 11 were obtained at 450 rpm speed. Curves given for 1 and 2 bars charge pressure have increasing slopes however the curve given for 3 bars has a decreasing slope. Distribution of data points

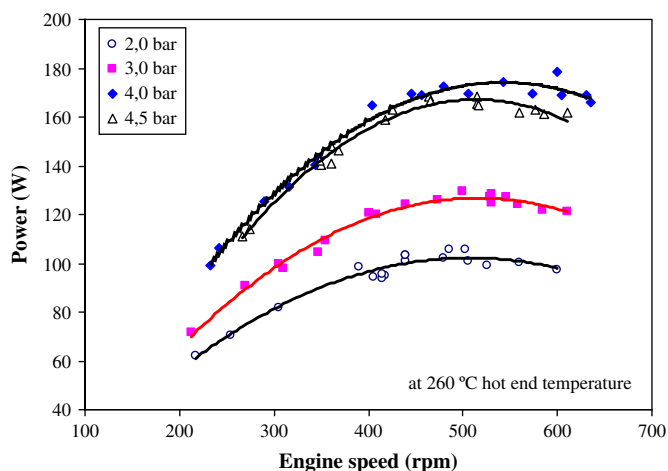


Fig. 9. Variation of engine power versus the speed at 260 °C.

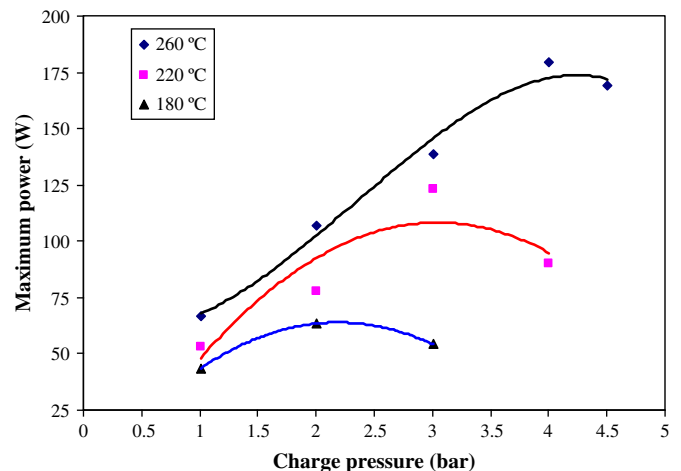


Fig. 11. Variation of maximum power versus the charge pressure.

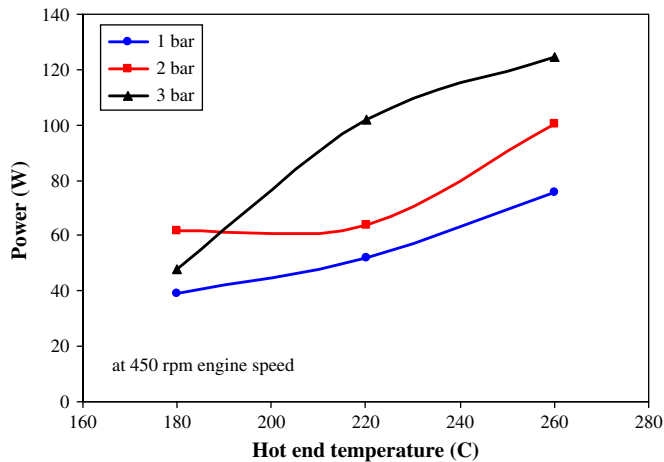


Fig. 12. Variation of power with hot end temperature at 450 rpm constant speed.

implies that there is a linear relation between the hot end temperature and power.

5. Conclusions

The working fluid was charged into the engine block and no leak was observed up to 5 bars. The initial running hot end temperature of the engine was measured as 118 °C for 1 bar charge pressure of working fluid which is helium. The maximum hot end temperature reached with a displacer cylinder without augmented external surface was measured as 260 °C. The maximum torque and power, determined as 3.99 Nm at 266 rpm and 183 W at 600 rpm respectively, were obtained at 260 °C hot end temperature and 4 bars charge pressure. The effect of charge pressure on the performance characteristics is positive up to a certain value and then negative. The relation between the hot end temperature and power seems to be linear. For low and moderate heat sources including solar energy the engine promises a reasonable performance.

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